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NOISE DIAGNOSTIC STRATEGIES IN A FRACTIONAL HORSEPOWER RECIPROCATING PISTON COMPRESSOR

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ABSTRACT

A diagnostic test program to evaluate the noise sources and transmission paths of a fractional horsepower reciprocating piston compressor is presented. The proposed test program consists of three phases: repeatability and controllability tests, exterior sound field measurements, and noise source identification.

In the first phase a controllable and repeatable test is devised using a fully-condensing load stand because of its increased thermal stability over the hot gas load stand. Repeatable measurements are necessary for meaningful comparisons between measurements. The ability to repeat sound pressure measurements with the load stand within 3 dB is generally acceptable.

The second phase involves measurement of the exterior sound field to determine the frequencies at which noise problems exist. Directivity measurements of the noise field is also performed to establish the directional characteristics of the noise radiation. Sensitivity testing of the compressor to its operating parameters (e.g., suction temperature, suction pressure, discharge pressure, and running speed) is performed to establish how each operating parameter affects the noise generation mechanisms, and to identify the frequencies of the structural and acoustic resonances of the compressor.

The final phase of the noise testing procedure involves determining the sources of the noise problems. An impact test of the compressor shell gives the natural frequencies and mode shapes of the shell, verifying the structural resonances. Similarly, a simple acoustic "mode analysis" identifies the character of the acoustic cavity modes.

INTRODUCTION

Fractional horsepower reciprocating compressors are commonly found in HVAC systems such as household refrigerators and air conditioners. The periodic operation of the compressor and the noise radiated during operation has been deemed to be a considerable problem by the consumer, especially during the quiet hours of the night. As a result, there is an increased need for understanding the noise and vibration characteristics of these compressors for the purpose of noise control.

In reciprocating piston compressors, the compression process is the origin of all noise radiated from the compressor. Suction gas pulsations, compressor mechanism vibrations, and discharge gas pulsations are all caused by the compression process and are the main noise sources in the compressor. Figure 1 shows a typical noise path flow chart for a reciprocating piston compressor. The noise can be transmitted through three paths, the lubricating oil path, the refrigerant gas path, and the mechanical path. All noise paths include the compressor shell causing either direct noise from the compressor, or indirect noise because of the excitation of the surrounding structure (refrigerator) by the compressor.

Noise control for any mechanical system can be achieved by one of three methods: treating the noise source, treating the transmission path, or treating the receiver of the noise (i.e., the person hearing the noise). The third option is not practical so special attention is paid to investigating the sources of the noise and the paths through which they are transmitted.

The objective of this paper is to provide a general noise control test procedure which can be implemented to investigate the dominant noise generation mechanisms and transmission paths in fractional horsepower reciprocating piston compressors.

DEVELOPMENT OF A CONTROLLABLE AND REPEATABLE TEST

The Load Stand

In beginning a compressor noise test, it is necessary to devise an experimental setup which can be easily controlled, and is able to produce repeatable operating conditions and measurements. To accomplish this a load stand is built to simulate the refrigerant cycle seen by the compressor. The method used to simulate operating parameters at the suction and discharge sides of the compressor is not critical. There are many ways of accomplishing this, the two most common are the hot-gas load stand, and the fully-condensing load stand. They both provide a load to the compressor, and the correct thermal conditions seen by the compressor, but they differ in the method which this is accomplished.

The refrigerant gas, R-12, in a hot-gas load stand is maintained in the superheated region. Thus, a hot-gas load stand uses only an evaporator and expansion valve to achieve the desired temperatures and pressures, all resulting in a system which is very quick to achieve the desired operating conditions since no condensing is required. In contrast, the fully-condensing load stand utilizes a condenser to change the state of the superheated refrigerant gas to a saturated liquid before expanding the refrigerant and evaporating it back into a superheated vapor. Figure 2 shows the thermal differences between the two cycles. The advantage of the fully-condensing cycle over the hot-gas cycle is improved stability due to added thermal momentum in the condensing cycle. For this reason a fully-condensing load stand is used in the present investigation.

Repeatability Measurements

The repeatability of the noise measurements is affected by three components in the experimental setup: the compressor itself, the load stand, and the measurement equipment. It is difficult to evaluate the variability of these three factors individually, so a study is made on the overall repeatability of the measurement system. It is necessary that the repeatability be evaluated to be able to compare measurements made at different times.

Five sound pressure measurements on each of 3 days are made, performing the standard start-up and shut-down procedure before and after each measurement. It was determined that the entire system was repeatable with a standard deviation of 2.5 dB for the same day measurements and 3 dB for all the measurements for the three day period.

It is necessary to establish the overall repeatability of the measurements in order to evaluate the effects of a design modification. If a reduction of the sound pressure level results after a design modification is made, it must be known that this reduction is a result of the modification and not due to random variations in the measurement system. Thus, any design modification which results in a sound pressure level change exceeding 3 dB is due to the design modification. Conversely, the reason for a sound pressure level change of less than 3 dB cannot be established with any certainty. It may be due to the modification and/or the random variations of the measurement system.

DETERMINATION OF THE EXTERIOR SOUND FIELD

In order to determine the particular frequencies at which most of the noise radiates, a characterization of the external sound field needs to be made. Microphones equally-spaced around the compressor are used to measures the narrow band and 1/3 octave band noise spectrum radiated from the compressor when run at its specified operating conditions. These measurements are made in an anechoic environment to minimize the effects of the surroundings on the compressor noise field. Care is also taken to ensure that the noise measurements were being acquired in the far field, so as to eliminate any near field effects.

Directivity Measurements

The directivity of the noise radiated by the compressor is evaluated by measuring the sound pressure level at locations around two horizontal planes and two vertical planes around the compressor. These measurements allow a determination to be made regarding the directional nature of the noise. If noise spectra measured at different locations are similar, i.e., each exhibits similar noise characteristics, then fewer microphones are needed to accurately describe the noise field, thus reducing the amount of data which needs to be collected and analyzed.

Sensitivity Testing

With the directivity of the sound field established, sensitivity testing of the compressor to its operating conditions can be performed. The sensitivity tests involve varying one of the compressor operating parameters while holding the remaining operating parameters constant and measuring the radiated sound to test the effect of the parameter on the noise generation mechanisms of the compressor. The sensitivity of the compressor to small changes in the suction temperature, suction pressure, discharge pressure, and running speed of the motor is tested to determine how each affects the noise generation mechanisms of the compressor.

Of these tests the running speed sensitivity test is the most enlightening. This test enables the pumping harmonics to be shifted through a range of frequencies allowing a determination to be made regarding the different structural and cavity resonances of the compressor. When a pumping harmonic nears a resonance of the system, the gas pressure pulsations reinforce themselves, increasing the sound pressure level until it reaches a peak right at the point where the pumping harmonic coincides with the resonant frequency as illustrated in Figure 3. After passing through the resonance the sound pressure level of this harmonic decreases, while the sound pressure level of the next lower harmonic similarly begins to increase with its proximity to the resonance.

The sensitivity tests also indicated that the test compressor is very sensitive to the suction pressure. The sound pressure level of the harmonics of interest generally increase monotonically with increasing suction pressure due to the added gas pressure pulsations at the higher suction pressures (see Figure 4a). Figure 4b illustrates that there is little change in sound pressure level due to a change in the suction temperature, and Figure 4c indicates that the compressor is insensitive to slight changes in discharge pressure. Generally, the sound pressure level will only vary significantly with temperature near a gas resonance. Also, the sound pressure level will generally not vary with discharge pressure due to the fact that the discharge pressure pulsations are effectively contained by the discharge muffler and discharge line.

NOISE SOURCE IDENTIFICATION

The final phase of the noise testing procedure is to identify the major sources of the compressor noise. The sensitivity testing gave an indication of the problem frequencies in the compressor. With this knowledge, a determination can be made regarding the nature of the resonance, either structural or cavity. A structural resonance occurs when a primary dimension of the compressor shell corresponds to an integral number of wavelengths (or sometimes half-wavelengths) of the bending waves excited in the shell. Under these conditions the vibrational energy in the shell can reinforce itself causing a resonance condition. Similarly, an acoustic cavity resonance occurs when a fundamental dimension of the compressor cavity (transverse or annular) corresponds to integral number of half-wavelengths (transverse), or full wavelengths (annular) of the excitation frequency. Generally, structural resonances occur at frequencies above 1,000 Hz due to the highly stiffened nature of the oval-shaped compressor shell. Conversely, acoustic cavity resonances generally occur at frequencies below 1,000 Hz because the transverse and circumferential dimensions of the interior cavity of the compressor can more easily reinforce themselves in this frequency range. However, acoustic cavity resonances can also occur at high frequencies.

Structural Resonances

To identify the structural resonances of the compressor, an impact test of the compressor shell is done. The impact test consists of hitting the compressor shell at a point with a force hammer and measuring the response of the shell to this force with an accelerometer. This procedure is repeated many times at many points around the shell. Modal analysis software is used to calculate the natural frequencies of the compressor shell and display the natural modes of vibration. When one of the pumping harmonics of the compressor coincides with one of these shell natural frequencies, the shell resonates and noise is radiated very efficiently from the compressor.

Cavity Resonances

The cavity resonances of the compressor are identified by performing a simple acoustic "mode analysis" [1]. This involves measuring the three circumferential dimensions, and the three transverse dimensions of the interior compressor cavity indicated in Figure 5. The wavelength in the cavity is then calculated by

$$\lambda = \frac{c}{f} \tag{1}$$

where λ is the wavelength, c is the speed of sound of the refrigerant, and f is the resonant frequency. If any circumferential dimension coincides with a multiple of the calculated wavelengths a cavity resonance can be expected. Likewise, if any half-wavelength multiple coincides with a transverse cavity dimension a resonance is produced.

To verify this analysis, internal pressure transducers can be mounted through the shell wall at various locations around the compressor shell to measure the pressure pulsations around the cavity. With this information the acoustic cavity modes can be mapped out.

CONCLUSIONS

Since noise radiated by the compressor in a refrigerator accounts for a major portion of the total noise emitted from the refrigerator it is desirable to control the compressor noise. The first step in solving noise problems is to identify the sources and transmission paths of the noise. This paper outlines a general noise diagnostic program to investigate the noise characteristics of fractional horsepower reciprocating piston compressors.

To begin the investigation a controllable and repeatable test needs to be developed using either a hot-gas load stand or fully-condensing load stand to provide the correct thermal conditions to the compressor. The load stand developed is stable and provides repeatable test conditions, and repeatable sound measurements within 3 dB.

A determination of the directional characteristics of the noise radiated by the compressor needs to be performed in order to minimize the number of microphones required to get an accurate picture of the overall noise field of the compressor. An evaluation of the exterior sound field around the compressor gives insight into the problem frequencies of the compressor. The sources of these resonant frequencies can be determined by a modal analysis of the shell and a simple acoustic "mode analysis" of the compressor cavity.

Once the dominant noise sources and transmission paths are determined, noise control strategies can be identified to eliminate or reduce the noise emitted from these sources by either reducing the source level or modifying the transmission paths.

ACKNOWLEDGEMENT

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REFERENCE

[1] Hamilton, J. F., Measurement and Control of Compressor Noise. Short Course Notes. Ray W. Herrick Laboratories, Purdue University, (1988).

FIGURES

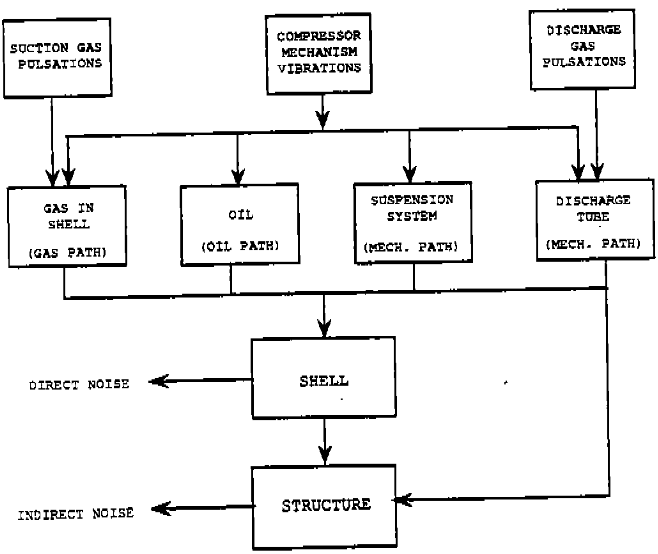

Figure 1: Block Diagram of Noise Paths in a Reciprocating Compressor

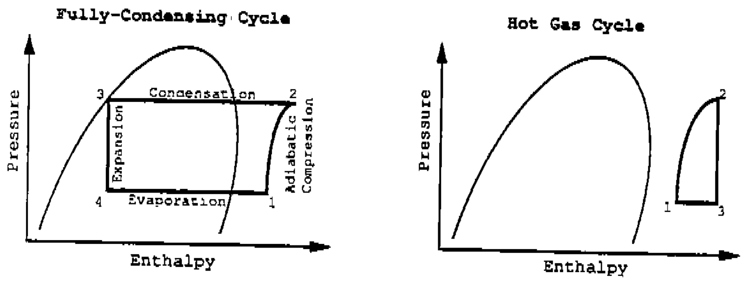

Figure 2: Pressure vs. Enthalpy Diagrams for Fully-Condensing and Hot Gas Thermal Cycles

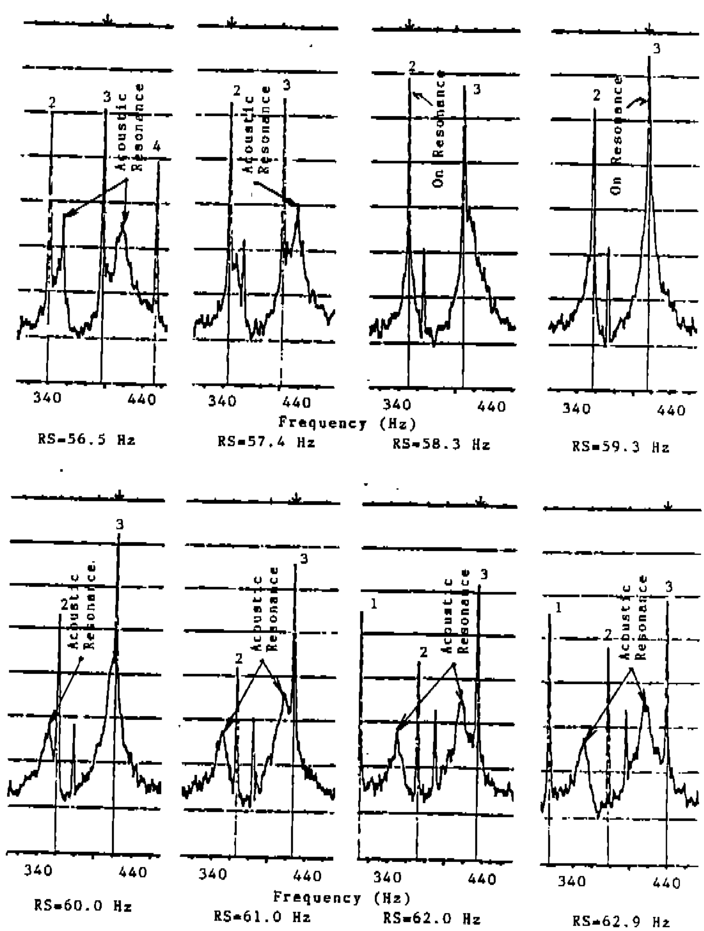

Figure 3: Shifting Pumping Harmonics Through Acoustic Resonances in the Running Speed Sensitivity Test (RS=Running Speed)

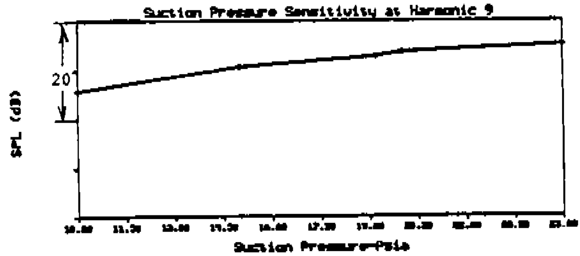

Figure 4a: Sound Pressure Level Sensitivity to Suction Pressure

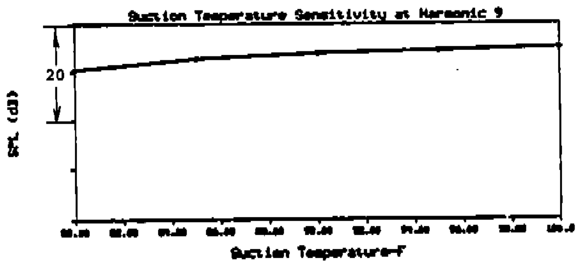

Figure 4b: Sound Pressure Level Sensitivity to Suction Temperature

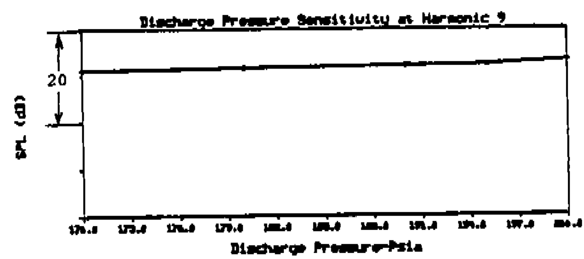

Figure 4c: Sound Pressure Level Sensitivity to Discharge Pressure

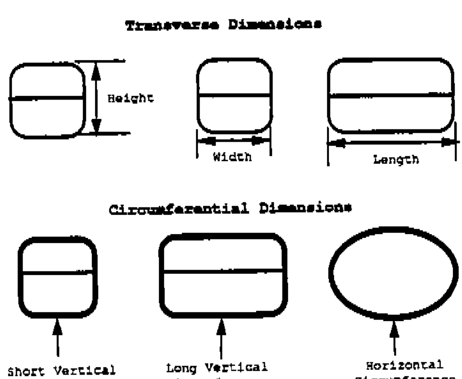

Figure 5: Dimensions Measured for Acoustic "Mode Analysis"